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PERFORMANCE EVALUATION OF CHLORINE FREE ZEOTROPIC REFRIGERANT MIXTURES IN HEAT PUMPS -COMPUTER STUDY AND TESTS-

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ABSTRACT

Fifteen binary zeotropic refrigerant mixtures consisting of the components R23, R32, R125, R134a, R143a, and R152a are investigated as possible replacement fluids for R22. The two mixtures of R32/R134a and R32/R152a showed COP improvements over R22 of up to 24% (depending on the operating condition and mixture composition) at the same capacity as with R22 while using counter flow heat exchange in evaporator and condenser. The use of a liquid line to suction line heat exchanger proved to be advantageous for both mixtures. The overall conductance for both mixtures is evaluated to be equal to or up to 22% greater (R32/R152a) than that of R22. Therefore, the heat exchanger size used with R22 should be sufficient to achieve performance increases with these zeotropic mixtures.

NOMENCLATURE

A	-	Area (m_2)
COP	-	Coefficient of Performance
d	-	Differential
GWP	-	Greenhouse Warming Potential
HCFC	-	Hydrofluorocarbon
LSHX	-	Liquid Line to Suction Line Heat Exchanger
MBHP	-	Mini Breadboard Heat Pump
NIST	-	National Institute of Standards and Technology
ODP	-	Ozone Depletion Potential
$\dot{Q}$	-	Heat Capacity (W)
s	-	Entropy (kJ/(kg*K)
T	-	Temperature (K)
UA	-	Overall Conductance (W/K)
w	-	Work (W)
x	-	Composition
ZRM	-	Zeotropic Refrigerant Mixture

subscripts

1,2,3,4	-	State Points in Figure 2
h	-	Heating
i	-	Counter, State Point in Cycle Figure 2
int. rev.	-	Internally Reversible, (also $|_r$)
m	-	Mean
n	-	Natural Number
net	-	Net Amount
r	-	Refrigerating

INTRODUCTION

To date, the commonly used refrigerant in residential heat pumps in the United States is R22. The GWP of R22 is 0.34 [1] and its ODP is 0.05 [1] (values are relative to R11). Currently, a production cap for this refrigerant is scheduled for the year 2015 [2], and a production phase-out is scheduled for the year 2030 [2]. As of now, there is no pure refrigerant or azeotropic mixture available that could be used to replace R22 without a significant performance decrease. The performance decrease is manifested by either a lower volumetric capacity or by a lower COP under the same operating condition as with R22. These shortcomings of pure fluids can be overcome if zeotropic refrigerant mixtures (ZRMs) are used. Their possible benefits have been described by Didion and Bivens [3]. Several research projects involving ZRMs [4], [5] have proven the potential benefits of zeotropic mixtures; however, the fluids used in these earlier studies have ODPs that are non-zero. Therefore, they are not acceptable as replacement fluids for R22.

The project described in this presentation outlines a new approach in determining possible ZRMs from a variety of non-ozone depleting pure refrigerants. Six non-ozone depleting HFC refrigerants were chosen as possible mixture components. They are the following chemical derivatives of methane and ethane: R23, R32, R125, R134a, R143a, and R152a. All fifteen possible binary mixtures of these candidates were simulated under typical heat pump conditions by means of the NIST developed "CYCLE11" [6] simulation program. Each mixture was investigated over its full composition range and the results compared to R22.

From these fifteen possible binary working fluids two mixtures, R32/R152a and R32/R134a, are predicted to perform better with respect to COP and volumetric capacity than R22 if counterflow heat exchangers are used. The COP of the R32/R152a mixture is predicted to be two to six percent higher than the R32/R134a mixture if no liquid line to suction line heat exchanger (LSHX) is used. These two ZRMs were then tested in the Mini Breadboard Heat Pump (MBHP) that was designed and built at NIST. The test apparatus is a water/glycol to water/glycol heat pump that uses a counterflow evaporator and condenser. The tests confirmed the findings of the simulation study: COP improvements of up to 24% (compared to R22) were measured with the mixtures at the same volumetric capacity as with R22. The implementation of a LSHX in the system increased the system performance for both mixtures. The R32/R134a mixture showed a larger performance increase than the R32/R152a mixture due to the LSHX implementation. Consequently, the performance advantage of the R32/R152a mixture over the R32/R134a mixture that was measured in the test apparatus in the operating mode without the LSHX could not be measured in the operating mode with the LSHX.

Using the test data it was possible to extract the overall conductance, UA, in the evaporator and the condenser. The UA-values obtained for both mixtures are 0% to 22% higher than the R22 values in the case of the R32/R152a mixture. For the R32/R134a mixture, the UA-values are equal to or up to 13% higher.

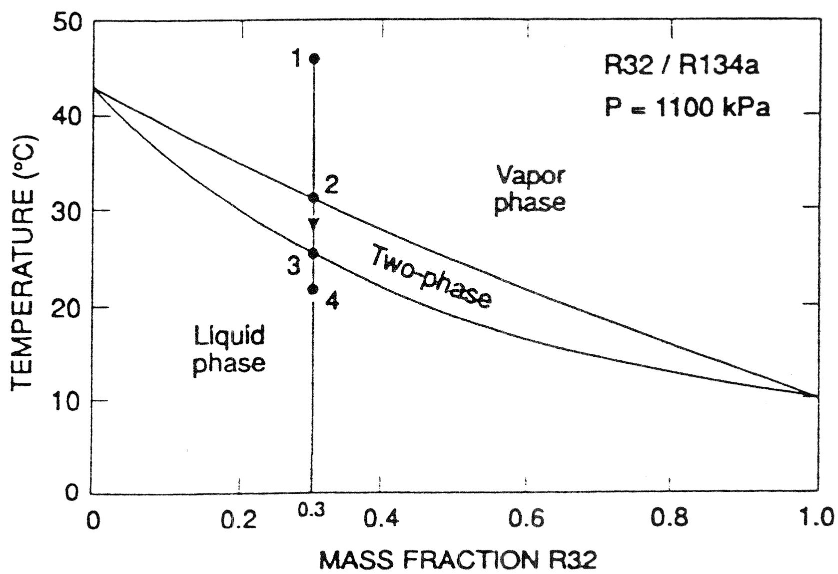

Figure 1 Temperature Glide of a Zeotropic Mixture

THEORY

In order to better understand the potential benefits of the usage of zeotropic[1] refrigerant mixtures over pure refrigerants as working fluids in heat pump cycles, it is useful to emphasize the differences between the two fluid systems and their commonly used ideal reference cycles.

The most obvious difference between a pure refrigerant and a zeotrope appears in their different

[1] zeotropic means the same as nonazeotropic but is preferred by the authors because its simplicity is more to the point (i.e., change in boiling) and eliminates the double negative prefix (i.e., "non" and "a").

phase change behavior. A pure fluid shows a constant evaporation or condensation temperature at a constant pressure. A ZRM, however, shows a change in evaporation and condensation temperature at a constant pressure. In the following, this behavior is referred to as the temperature glide (of evaporation or condensation). This temperature glide can be shown in a temperature-composition (T-x) diagram as can be observed in figure 1 for the binary mixture of R32 and R134a. The figure shows condensing at a pressure of 1100 kPa and an overall composition of 30% R32 and 70% R134a. The process in figure 1 can be described in three parts:
1: desuperheating vapor from point 1 to point 2,
2: condensing the fluid from point 2 to point 3, and
3: subcooling the liquid from point 3 to point 4.
The temperature glide that can be experienced for this particular mixture at this pressure is about 6 °C ($\approx$ 11 °F) (measured between points 2 and 3). The temperature glide which depends on the pressure, on the mixture composition, and on the fluids that form the mixture can be significantly larger. Figure 1 also shows how the phase change of pure fluids appears in such a diagram. For pure R32 (mass fraction equals one) and pure R134a (mass fraction equals zero), no temperature glide can be exhibited and therefore, the phase change appears as a single point.

The difference between the two kinds of working fluids explained in the previous paragraph leads to two different ideal reference cycles. The ideal presentation of a refrigeration cycle using a pure refrigerant as working fluid is usually performed by the Carnot cycle which is described by an isentropic compression, an isothermic condensation, an isentropic expansion, and an isothermic evaporation of the working fluid. This process is shown in the temperature-entropy (T-s) diagram in figure 3 together with the ideal reference cycle for the zeotropic mixture cycle which is the Lorenz cycle. A system schematic for both cycles is provided in figure 2. For the Lorenz cycle, the compression and expansion are considered isentropic as in the case of the Carnot cycle. The condensation and evaporation are isobaric processes, thus allowing for temperature glide, which is experienced by ZRMs during phase change at constant pressure. In figure 3, the Carnot cycle is described by the points 1C-2-3C-4 and the Lorenz cycle by the points 1L-2-3L-4. Under the assumption that the processes 2-3C, 2-3L, 4-1C, and 4-1L are internally reversible [7], it can be shown [8] that the condensation heat and evaporation heat is represented by the integration of the temperature with respect to the entropy as expressed in eq (1). For the ideal cycles, it is furthermore assumed that the heat exchangers are infinitely large so that heat transfer fluid temperatures can match those of the working fluids. This also requires an infinitely large heat transfer fluid flow rate for the Carnot system in order to provide for no temperature change on the heat transfer fluid side. As a next step, a first law analysis for the two systems that are represented by figures 2 and 3 is performed [eq (2)]. Equations (1) and (3) lead to the conclusion that the areas enclosed

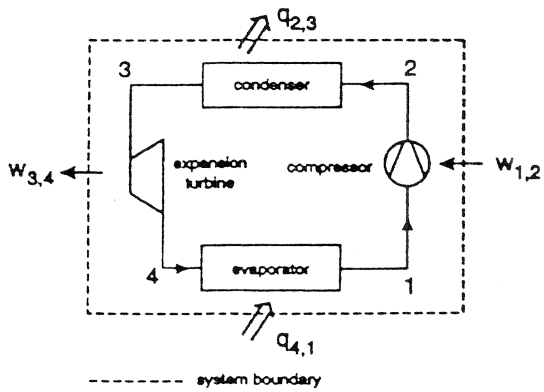

Figure 2 System Used for the First Law Analysis

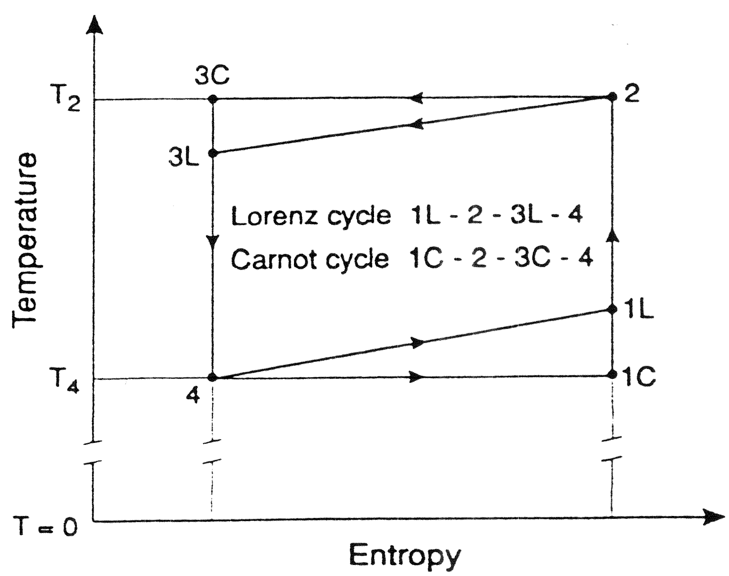

Figure 3 Ideal Reference Cycles for Zeotropic Mixture (Lorenz cycle) and Pure Refrigerant (Carnot Cycle) Heat Pumps

by the lines describing each cycle in the T-s diagram (Fig. 3) represent the work that is needed to drive each process under the defined assumptions.

$$q_{tot,rev.} = q\big|_r = \int T\,ds \tag{1}$$

$$0 = \sum_i q_i\big|_r + \sum_i w_i = q_{4,1}\big|_r - q_{2,3}\big|_r + w_{1,2} - w_{3,4} \tag{2}$$

Now using eq (1) and rearranging yields:

$$w_{1,2} - w_{3,4} = w_{net} = q_{2,3}\big|_r - q_{4,1}\big|_r = \int_2^3 T\,ds - \int_4^1 T\,ds \tag{3}$$

The work for the Lorenz cycle is smaller than for the Carnot cycle while both cycles result in the same achievable maximal (condenser) and minimal (evaporator) heat transfer fluid temperatures T_2 and T_4 if counterflow heat exchange is used for the Lorenz cycle. Using figure 3 and eq (1), it can also be concluded that the maximum achievable refrigerating COP (COP_r) for the ideal Lorenz cycle [eq (4)] is greater than for the Carnot cycle. This can be concluded since the area representing the work is smaller and the area representing the cooling capacity is greater for the Lorenz cycle. This fact is not so obvious from the area point of view for the heating COP (COP_h) since the area representing the heating capacity is smaller for the Lorenz cycle. However, since the $COP_h = COP_r + 1$ [eq (5)] for both ideal cycles, it is shown that the cycle performance of the theoretical Lorenz cycle is also better in the heating mode. Thus, using a zeotropic mixture as working fluid in a heat pump cycle proves theoretically advantageous compared to a single component working fluid.

$$COP_r = \frac{q_{4,1}}{w_{net}} \tag{4}$$

$$COP_h = \frac{q_{2,3}}{w_{net}} = \frac{q_{4,1} + w_{net}}{w_{net}} = \frac{q_{4,1}}{w_{net}} + 1 = COP_r + 1 \tag{5}$$

SIMULATION

Determination Of The Operating Conditions

The simulation of the heat pump cycle was performed with the CYCLE11 [6] program that was developed at NIST. The heat transfer fluid temperatures were deduced with the help of the ASHRAE standard ANSI/ASHRAE 116-1983 [9] and tests with the MBHP for four operating conditions. The airflow rates in the

Table 1: Heat Transfer Fluid Temperatures Used for the Computer Simulations and Tests

Location	Test A[1] °C (°F)	Test B[2] °C (°F)	Test C[3] °C (°F)	Test D[4] °C (°F)
Condenser Inlet	35.0 (95)	27.8 (82)	21.1 (70)	21.1 (70)
Condenser Outlet	43.2 (110)	37.4 (99.3)	32.5 (90.5)	28.1 (82.6)
Evaporator Inlet	26.7 (80)	26.7 (80)	8.3 (47)	-8.3 (17)
Evaporator Outlet	14.4 (58)	13.8 (56.8)	2.7 (36.9)	-11.3 (11.7)

indoor and outdoor unit were assumed to be 0.05219 $m^3/(s\ kW)$ (400 scfm/ton) and 0.10438 $m^3/(s\ kW)$ (800 scfm/ton), respectively, thus, together with the inlet temperatures from the ASHRAE standard, the heat transfer fluid outlet temperatures were calculated for the high temperature cooling mode (Test A) as the layout condition. Since the system performance changes with the changing operating conditions in tests B, C, and D, the authors chose to use the established test condition A for test runs with R22 in the MBHP. The MBHP is a water/glycol to water/glycol heat pump. The heat exchangers were used in counter-flow mode for all tests in this study. After establishing the heat transfer fluid mass flow rates in the heat exchangers for test condition A,

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[1] high temperature cooling
[2] low temperature cooling
[3] high temperature heating
[4] low temperature heating

these flow rates were kept constant with respect to the outdoor and indoor units for the other three operating conditions (low temperature cooling, high temperature heating, low temperature heating). With these flow rates and the heat exchanger inlet temperatures from the ASHRAE standard it was possible to establish reasonable heat transfer fluid in- and outlet temperatures for condenser and evaporator by running tests with the MBHP. The resulting operating temperatures are listed in table 1 and were used for all working fluids in the simulation calculations.

Mixture Components and Selection Criteria

The main requirement for the refrigerants considered as mixture components is that they do not deplete the ozone layer. At the same time the authors chose to consider only chemical derivatives of methane and ethane. These two criteria resulted in the consideration of six pure refrigerants as mixture components: R23, R32, R125, R134a, R143a, and R152a.

The mixtures themselves that could replace R22 have to meet other criteria such as performance, environmental, and engineering criteria. The ODP of any of the fifteen possible mixtures is zero. The GWP of an eventual mixture should be lower than that of R22. For safety reasons the mixture should also be nonflammable and nontoxic. The discharge temperatures should always be lower than 150 °C (302 °F) and the discharge pressures should not exceed 2600 kPa (377 psia). At the same time, the performance requirements of the mixtures are that the same or higher COP is achieved at the same or higher volumetric capacity than with R22. Table 2 lists the investigated refrigerants with some important property data.

Table 2: Property Data of the Investigated Refrigerants

refrigerant	critical point values [10]		NBP [10]	ODP [1]	GWP [1],[11]	toxicity [12],[13]	flammability [12],[14]
	temperature (°C)	pressure (kPa)	(°C)	(/)	(/)	(/)	(/)
R22	96.15	4988	-40.85	0.05	0.34	low	no
R23	25.83	4820	-82.05	0.0	21	low	no
R32	78.41	5830	-51.75	0.0	0.13	low	yes
R125	66.25	3631	-48.55	0.0	0.58	low?	no
R143a	73.10	3811	-47.35	0.0	0.74	low?	yes
R134a	101.06	4056	-26.15	0.0	0.26	low?	no
R152a	113.29	4520	-24.15	0.0	0.03	low	yes

Simulation Results

The simulation was performed without compressor speed variations and without the implementation of the LSHX. Each of the fifteen mixtures was simulated over the whole composition range in steps of five percent for all four operating conditions.

Out of the fifteen binary mixtures investigated in this research, there are two binary mixtures that indicate a better performance than R22: R32/R152a and R32/R134a. Figures 4 and 5 show the COP and volumetric efficiency results for the low temperature cooling condition. Although the pure components violate at least one of the established selection criteria (see above), wide ranges of mixture compositions perform better in COP and show a higher volumetric capacity while being acceptable from the engineering aspects.

The mixture of R32/R152a is flammable since both pure substances are flammable (Table 2). Both pure refrigerants appear to have low toxicities. The GWP of this mixture is the lowest of all possible binary combinations. The calculations for R32/R134a show a smaller performance improvement than for R32/R152a but still a significant increase compared to R22. Both R32 and R134a are in the class of low toxicity and only R32 is flammable. Flammability tests indicate that this mixture is flammable at room temperature for R32 mass concentrations above 56% [15]. The possible different compositions of the vapor phase and the liquid phase during an equilibrium phase change of ZRMs require the overall composition of the mixture to be lower than the 56%-mass limit in order to ensure that an eventual leakage composition is not flammable. From a worst case leakage analysis it can be concluded that a mixture with an overall mass concentrations of 30% R32 should not be able to leak a flammable mixture for operating pressures above 200 kPa (29 psia) which relates to a saturation temperature of ≈-26 °C (≈-15 °F). The GWP of R32/R134a is higher than that of R32/R152a but still significantly lower than that of R22.

Given the results of the computer study, the authors chose to conduct tests with these two refrigerant mixtures within certain composition ranges. The composition ranges of the tested mixtures are largely determined by the attempt to find a mixture that achieves the same volumetric capacity and the same COP as R22 under all operating conditions. At the same time, excessive amounts of R32 should not be used in order to provide for acceptable discharge pressures even under extreme operating conditions. Therefore, the authors chose to run tests for both refrigerant mixtures in a range between 15%-mass and 40%-mass R32. For all these compositions, the pressures and temperatures are expected to be well within the acceptable range.

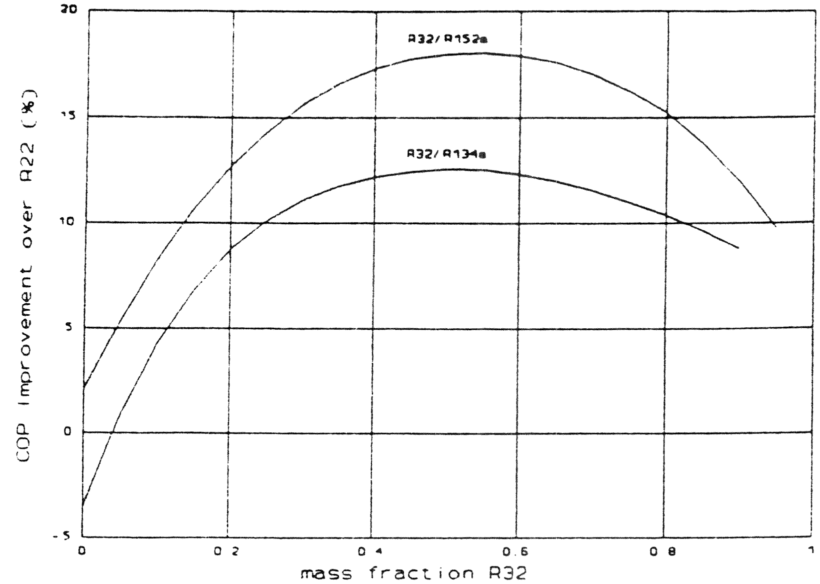

Figure 4 Relative COP of R32/R134a & R32/R152a; low temperature cooling conditions

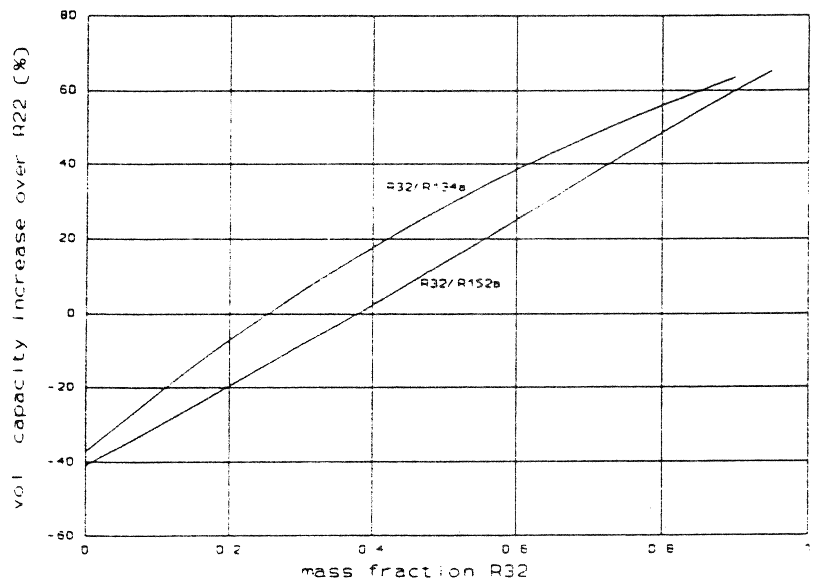

Figure 5 Relative volumetric capacity of R32/R134a & R32/R152a; low temperature cooling condition

TESTS

The tests were performed with the MBHP using counter flow heat exchange. In order to allow for a fair comparison of the mixtures with R22, the tests were conducted at the same capacity as achieved with R22 under the same operating condition. This test criterion is an attempt to use a method to compare mixtures with pure refrigerants that was proposed by McLinden and Radermacher [16] .

The measured COPs for the tested composition range are presented in figures 6 and 7 for the high temperature cooling condition for both mixtures. Figure 6 shows the tests without LSHX and figure 7 shows the test series with LSHX. In order to fulfill the capacity and COP requirements, the R32/R152a mixture should consist of at least 50%-mass R32. This high fraction of R32 is necessary to ensure a capacity that meets the R22 capacity for all operating conditions. The significant condition for the capacity is the low temperature heating test [17] . At all other operating conditions, a significant increase in COP can be expected with this composition given counterflow heat exchange in a constant compressor speed system. The operating pressures and temperatures indicate no problem in the usage of this mixture. In fact, the discharge temperature should be about 5 K lower than with R22 if a suction side cooled compressor is used. This is of significance for application in which

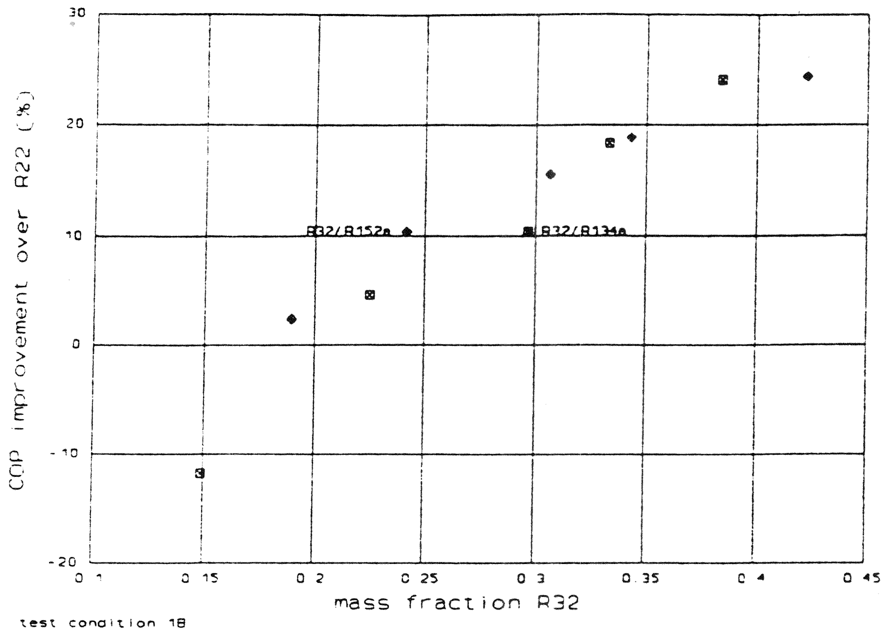

Figure 6 Relative Cooling COP; High Temperature Cooling Condition (1B)

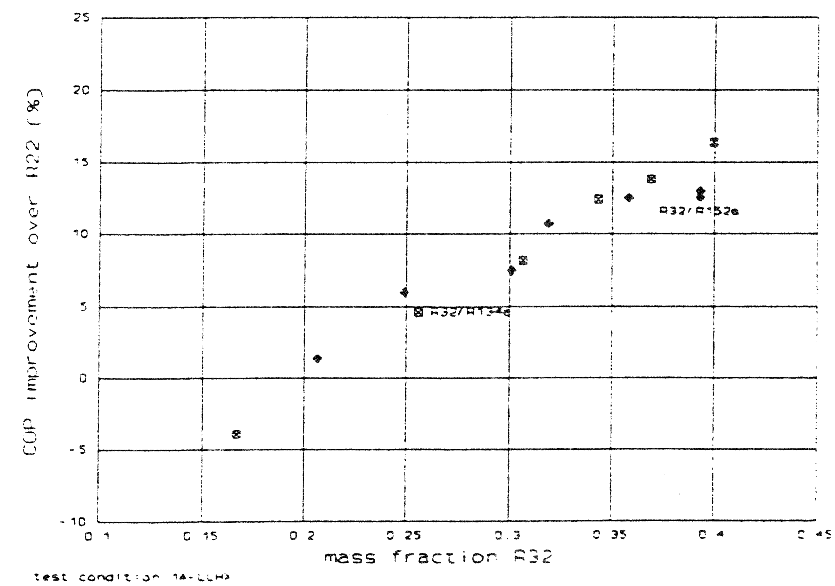

Figure 7 Relative Cooling COP; High Temperature Cooling Condition with LLHX (1A-LLHX)

typically R502 is used to lower the discharge temperatures. The R32/R152a mixture has the lowest GWP possible of the tested mixtures which is about one-fourth of the R22 value. However, the mixture is flammable in the whole composition range.

The other ZRM investigated has been found to perform not as well as the R32/R152a mixture in the region of lower R32 content for the cooling tests if used without LSHX. But, for mixtures consisting of at least 35%-mass of R32, the R32/R134a zeotropic mixture shows equivalent or better performance than the R32/R152a mixture for all operating conditions especially if a LSHX is used in the system. There are two definite advantages of this mixture. The performance in the heating mode is better than that of R32/R152a since, in general, this mixture has a higher volumetric capacity at the same R32 concentration. This is important since it affects the need for supplementary heat (resistance heating) during the heating period. The second advantage, and this might be even more important, is the flammability aspect. From the current knowledge about the flammability limit, it is concluded that R32 contents of about 30 to 35% should be tolerable.

Interesting to note is that the computer simulations indicated a better performance for the R32/R152a mixture over the whole composition range. This is not found to be the case for the test results. This deviation from the computer prediction can be attributed to the differences in operating parameters such as pressure drops, compressor efficiency, etc. which are not constant as was assumed for the simulation runs.

The tests with LSHX favor the mixture of R32/R134a compared to the R32/R152a mixture since they show a higher increase in COP when compared to the tests without the LSHX. The possible benefits of the LSHX are remarkable with respect to another aspect. As a separate counterflow unit in the refrigeration cycle, the LSHX impact on performance is independent of the kind of evaporator or condenser used in the system. If these increases in COP with the usage of the LSHX can be validated for other test conditions, then there are two new aspects. The first one is that with the same R32 content, higher COPs can be achieved. The other aspect is that since the flammability is an important issue, the R32 content could be lowered to a point where the volumetric capacity is still satisfactory. That would require only 30%-mass R32 in the mixture instead of 35%, at which point this mixture should be safely usable.

The increases in COP over R22 that were measured with both mixtures amount to up to 24% (test condition B; about 40%-mass R32, no LSHX). These high increases, however, cannot be expected with cross flow or parallel flow heat exchangers and with a design resulting in the same pressure drops as for R22. Nevertheless, the significant improvements that were measured offer enough potential so that cross flow or parallel flow heat exchange (as used in household heat pump units) should benefit from the usage of these mixtures under most operating conditions. Considering the use of a LSHX for the mixture of R32/R134a, it is very likely that an increase in COP remains even with heat exchangers that do not use the temperature glide of the mixtures (i.e., cross flow or parallel heat exchangers). This is the case, since a LSHX in counterflow can always be incorporated in a system. The advantage due to the implementation of the LSHX in a system is estimated to be about five percent for the R32/R134a mixture (comparing at the same mixture composition with and without LSHX).

The test results for all mixtures and compositions were obtained using the same test apparatus. There was no optimization of the test equipment with respect to pressure drop, compressor efficiency, heat exchanger surface area, etc. for any specific working fluid. This is important because the pressure drop in the heat exchangers for the R32/R134a mixture is significantly higher than that of the R32/R152a mixture (compared at the same R32 mass fraction). This pressure drop is system dependent, not refrigerant dependent. The compressor, however, does not differentiate operating pressures that are created due to pressure drops or due to the fluid properties. Thus, the pressure difference that the compressor has to overcome is increased due to the pressure drops in condenser and evaporator. For a real system the pressure drop is a design criterion. Therefore, for both mixtures the same pressure drops can be expected for a proper system layout.

Overall Conductance of the Heat Exchangers

The collected test data from the MBHP tests was used to estimate the overall conductance of the heat exchangers, UA, according to eq (6) [17]:

$$UA = \frac{\dot{Q}}{\sum_{i=1}^{n} \frac{A_i}{A} * \Delta T_{m,i}} \tag{6}$$

The overall conductance is chosen to represent the heat transfer coefficient on the refrigerant side, where the "U"-value is the mean heat transfer coefficient of the heat exchanger. This presentation is possible, since neither the heat transfer area nor the heat transfer fluid flow rate and heat transfer fluid temperatures change for each distinct test condition. "A" is the total heat transfer area, "$\dot{Q}$" is the transferred heat, "A_i" is the heat transfer area of section i, "$\Delta T_{m,i}$" is the logarithmic mean temperature difference of section i, and "n" is the number of sections that are used to calculate the $\Delta T_{m,i}$'s,. The measured refrigerant and heat transfer fluid temperatures were used to obtain the $\Delta T_{m,i}$ values. The results for the low temperature cooling

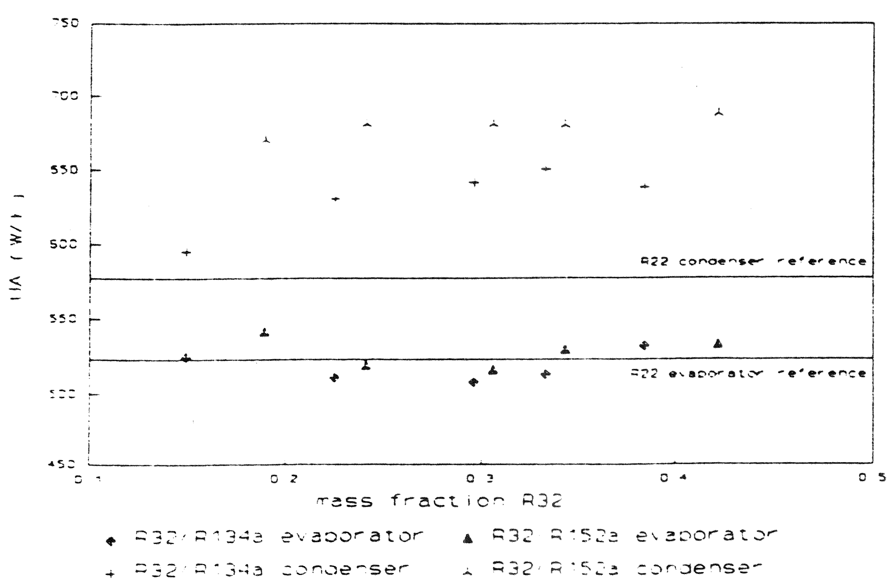

Figure 8 UA-Values for Low Temperature Cooling Test (1B). (extracted from test data)

condition are presented in figure 8 for both mixtures as a function of the R32 content together with the reference values of the R22 tests.

The condenser UA-values are found to be higher than in the evaporator. Under all operating conditions, it is found that in the condenser the mixture heat transfer coefficients are higher than those for R22. For the evaporator, the mixture heat transfer coefficients are very close to the R22 values. The R32/R152a mixture shows higher UA-values than the R32/R134a mixture in the condenser and almost identical values in the evaporator. The R32/R134a mixture tests show an improvement over the R22 UA-values by 3% to 13% in the condenser. The evaporator UA-values for this mixture vary from 3% smaller to 5% greater than with R22. The R32/R152a mixture shows higher UA-values than the R32/R134a mixture. In the condenser, a UA-value improvement of 10% to 22% over R22 is obtained. In the evaporator, the UA-values are between one percent smaller and 10% greater than for R22. The varying improvement ranges are caused by the different operating conditions and mixture compositions.

CONCLUSIONS

This study shows that the two zeotropic refrigerant mixtures, R32/R134a and R32/R152a, may be considered as replacements for R22 if the appropriate mixture compositions are chosen. Data indicate that multiple tradeoffs exist in the mixture performance for different system compressor speeds and mixture compositions.

The improvements of the R32/R152a mixture over R22 range from 24 percent for the low temperature cooling mode to two percent for the low temperature heating mode. At the same speed and capacity as R22, the mixture performs about 14% better in the low temperature cooling test and equal to R22 in the low temperature heating test. Operating pressures and temperatures of this mixture are well within acceptable limits. The Global Warming Potential of the tested mixture is about one-fourth the value of R22. However, this zeotropic mixture is flammable in the whole composition range.

R32/R134a appears to be the better choice of the two mixtures although the simulations do not predict it to perform as well as the R32/R152a mixture for all compositions. The test results indicate that for mixtures containing more than 35%-mass of R32, the performance is as good as that of the R32/R152a mixture without using the LSHX. In the heating mode, slightly higher COP's were measured compared to the R32/R152a mixture at the same R32 mass fractions. The improvements over R22 range from 24% in the low temperature cooling mode to six percent in the low temperature heating mode. If compared at the same speed and capacity as the R22 tests then the performance improvement over R22 is about four percent in the low temperature cooling mode. In the low temperature heating mode, the COP is about equal to that of the R22 tests. For R32/R134a mixtures with less than 30% to 35%-mass R32, this mixture may not pose a flammability risk if used for the heat pump application. The test results for the R32/R134a mixture show no problems with respect to extreme pressures or temperatures in the tested composition range. If the LSHX is implemented in the system then the R32/R152a mixture does not show any performance advantage over the R32/R134a mixture and the R32 content can be lowered to about 30%-mass, thus further lowering the flammability risk of the R32/R134a mixture.

The COP increases achieved for both mixtures offer a good performance increase potential so that even cross flow heat exchange systems may benefit from the usage of these mixtures. This is especially true for the R32/R134a mixture considering the use of a LSHX.

The test results of the mixtures are compared to those of R22 at the same heating/cooling capacity. All results were achieved with the same test apparatus, meaning there is no optimization with respect to system pressure drops, compressor efficiency, etc. for the different working fluids.

Both tested mixtures show overall conductances that are equal to or up to 22% greater (for R32/R152a; up to 13% greater for R32/R134a) than those of refrigerant R22 (higher in the condenser and equal in the evaporator).

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